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PROVISIONAL APPLICATION COVER SHEET

This is a request for filing a PROVISIONAL APPLICATION under 37 CFR 1.53(b)(2).

Docket Number	65,158-005	Type plus sign (+) inside this box —►	+

10/16/98

JC784 U.S. PTO

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JC541 U.S. PTO
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TITLE OF THE INVENTION (280 characters max)

ALL GEAR INFINITELY VARIABLE TRANSMISSION

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ENCLOSED APPLICATION PARTS (*check all that apply*)

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METHOD OF PAYMENT (*check one*)

☒ A check or money order is enclosed to cover the Provisional filing fees ☐ The Commissioner is hereby authorized to charge filing fees and credit Deposit Account Number: ______	PROVISIONAL FILING FEE AMOUNT ($)

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Respectfully submitted,

SIGNATURE ______ Date 10/16/98

TYPED or PRINTED NAME Samuel J. Haidle REGISTRATION NO. 42,619
(*if appropriate*)

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PROVISIONAL APPLICATION FILING ONLY

60/104477

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ALL GEAR INFINITELY VARIABLE TRANSMISSION

Field of the Invention

This invention relates in general to an all gear infinitely variable transmission.

Background of the Invention

Prototype vehicle transmissions incorporating the function generators of both Canadian Patent nos. 1,209,375 and 1,209,376, and the spiral type one-way clutches of United States Patent no. 4,341,294 have successfully demonstrated the viability of all-gear infinitely variable technology. However, the maximum kinematic range of bevel/bevel and bevel/carrier type function generators have in the past been limited to either the positive or negative periods of constant amplitude being generated. The kinematic range has been limited because of a limit to the kinematic ratio of the incorporated differential-gear-sets and a unidirectional characteristic of the spiral type one-way clutches. A further limitation of the prototype transmissions has been that translated energy is unidirectional and consequently engine drag is non-effectual.

A main disadvantage of the prior art infinitely variable all-gear transmissions is that the range of energy translation is restrained by either a kinematic limitation of the differential gear-sets or of the spiral type one-way clutches utilized in their respective function generators. A bevel/bevel type function generator of Canadian Patent no. 1,209,375 is limited in translation range to about forty percent (40%), while that of a bevel/carrier type function generator of Canadian Patent no. 1,209,376 is limited to about one hundred percent(100%), included infinite translation in some layouts. Other disadvantages in manufacturing and in particular of assembly make them excessively expensive.

Objects of the Invention

The disadvantages of the prior art may be overcome by providing engine braking and simultaneously increasing the maximum kinematic range of a coplanar loop type function generators by coupling both positive and negative periods of constant amplitude with the substituted bi-directional full compliment couplings that are made to uncouple to a passive mode by impinging linkage that is activated by shadow cams affixed to the two common phase shifteble driver gear elements of the variable ratio gear-sets of the generators during all times when amplitude varying periods of constant velocity are not being generated.

It is desirable to provide a method for generating constant contact ratio teeth flanks for variable ratio gear-sets with driven/driver ratios of respective pitch circles of congruency of constant relative rotational acceleration from 180° to 220° and more, with maximum amplitudes of driven/driver ratios from 0.1667 to 0.5000 and with sinusoidal return over the supplementary angle of rotation at constant acceleration.

It is desirable to provide protrusions of short durations on the pitch circles of the variable ratio gear-sets in the regions of transition from constant relative acceleration to sinusoidal return and vice versa, providing torque entrapment during periods of energy transfer in the function generators which can be momentarily suppressed as disengagement and/or engagement of the passive couplings takes place, thereby minimizing if not eliminating any shock loading on the couplings.

It is desirable to provide a twin, circular shadow cam actuator with one affixed to each of two separately rotatable common driver gears of the variable ratio gear-sets of the function generators. Radial displacement lobes on the twin circular shadow cams are of an arc length the equivalent of a supplementary angle of sinusoidal return rotation of the pitch circle profiles of the driver gears, so that, the duration of radial displacement of a common

cam follower to the twin circular shadow cams is subject to a relative phase relationship between them and the two affixed common driver gears. During in-phase rotation the lobes shadow each other and duration is the equivalent of a supplementary angle of sinusoidal rotation, but increases proportionally in length as the in-phase relationship of the two common driver gears of the variable ratio gear-sets changes in either a clockwise or a counterclockwise direction.

It is desirable to provide a twin shadow cam follower mechanism comprising of three rotating disc members, an axially fixed end member, an axially movable end member and a radially displaceable centre follower member incorporating a plurality of off-set rollers with conical ends that interface with off-set conical cavities on the inner surfaces of the end disc members, so that when all three discs rotate about a common axis the end discs abut against the centre disc, and when the center disc is made to rotate on an axis displaced from a common axis of end disc rotation when subject to the lobes of the shadow cams, interaction between the cone ends of the rollers and the conical cavities of the end discs causes the movable end disc to displace axially carrying with it slipper a striker linkage that disengages the conical, or cylindrical interfaces of the passive couplings.

It is desirable to provide a twin shadow cam follower mechanism comprising of a four sided, radially outward sliding member with a cam roller affixed to an inner end and with an angled surfaced outer end that abuts a similarly angled surface on a four sided, axially sliding member with common axis to a striker linkage that disengages the conical, or cylindrical slipper interfaces of the passive couplings during rotation when the cam roller is subject to the radial action of the lobes of the shadow cams.

It is desirable to provide a twin shadow cam follower mechanism consisting of a near 90^0 casing affixed rocker, with an arm extending axially with an incorporated cam roller so that when subject to the radial action of the lobes of a twin shadow cam a second

arm extending radially to the striker linkage disengages the conical, or cylindrical slipper interfaces of the passive couplings.

It is desirable to provide a hydraulically controlled rotary actuator to phase shift the common driver gears of the variable ratio gear-sets of the function generators of the subject infinitely variable transmission, with rotor thereof, together with a common driver gear and a cam of a twin shadow cam actuator, affixed to an outer concentric shaft; and with stator casing thereof, together with a second common driver gear and a second cam of the cam actuator, affixed to an inner concentric shaft. Within a cylindrical body of a hydraulic control valve affixed to the rotor, with integral ports extending to the clockwise and counterclockwise rotational cavities of the rotary actuator, is a cylindrical spool of the hydraulic control valve with free attachment to the stator inner shaft, with the axial position of the spool a consequence of oppositive variable hydraulic pressure force versus a spring force, so that when in balance, pressure and exhaust domains within spiral lands on the outer cylindrical surface of the spool are isolated from the respective integral ports of the stator by the spiral land boundary of the domains and the rotor and stator are held in an angular steady-state relationship. When a force unbalance exist, the pressure and exhaust domains are respectively exposed to the integral ports on separate sides of the spiral land boundary, with the result that hydraulic oil flow takes place to and from the directional cavities of the rotary actuator, forcing the rotor and stator to assume a different angular steady-state relationship. When the spool is forced to either respective end travel positions by a spring force or an overwhelming hydraulic forces, the radial displacement lobes of the tethered shadow cams are at minimum overlap and the period of rotational disengagement of the conical, or cylindrical slipper interfaces of the passive couplings are of maximum duration. Axial positioning of the spool valve is controlled by varying the hydraulic pressure to a spool end opposite that of the spring, with a varying pressure obtained from a pressure

chamber with inflow from a fixed orifice hydraulic pressure source and outflow controlled by a solenoid excited variable bleed needle valve, with pressure entrapment made to vary by a change in electrical pulse frequency on the solenoid, the higher the frequency the greater the entrapment.

It is desirable to provide an all gear rotary actuator to phase shift the common driver gears of the variable ratio gear-sets of the function generators of the subject infinitely variable transmission, comprising of a reverted-gear-train-loop with first and last gear elements, respectively attached to an outer and inner concentric shafts together with a common driver gear of the variable ratio gear-sets and a cam of a twin shadow cam actuator, a rotatable cage structure in which is bearinged a clustered second and third gear element respectively congruent with the first and last gear elements, so that, with a differential ratio between the first and second gear element than that of the third and last gear element, relative rotation of the cage about a common centre will effect a phase change between the first and last gear elements of the reverted-gear-train-loop and consequently the radial displacement lobes of the respective attached shadow cams that together with follower and striker mechanism disengage the conical, or cylindrical slipper interfaces of the passive couplings. With the ratio between the first and second gear elements of the loop greater than the ratio between the third and last gear elements, entrapped torque in the reverted-loop will tend to force a free cage to rotate at a faster speed than either the first or last gear elements. As a consequence their relative radial relationship will change in a particular order. When the cage is forced to rotate at a slower speed by a friction band mechanism, their relative radial relationship will change in an opposite order. Thus, if a particular order of change decreases the amplitude of translation thru the function generators and an opposite order of change increases the amplitude of translation, then variability of a subject infinitely variable transmission can be controlled by

selectively releasing the cage to either the entrapped torque forces of the reverted loop or to an external resistive torque of an all gear rotary actuator.

It is desirable to provide a self-positioning rotary actuator to phase shift the common driver gears of the variable ratio gear-sets of the function generators of the subject infinitely variable transmission, comprising of a flat sectioned radial coil spring respectively attached between an inner concentric shaft and a cylindrical flange of an outer concentric shaft together with respective common driver gear of variable ratio gear-sets and a cam of a twin shadow cam actuator, so that, either during pre-load torque entrapment or during maximum load torque entrapment on the radial coil spring, the lobes on the cylindrical cams are at minimum overlap and a maximum period is directed to the striker linkage that disengages the conical, or cylindrical slipper interfaces of the passive couplings. Accordingly, the action a self-positioning rotary actuator is kin to a torque converter, with a much more positive energy through put and with a much greater efficiency.

According to one aspect of the invention, there is provided a cage/pinion or annulus/pinion type function generators with incorporated full compliment type passive couplings having respective translation ranges of two hundred thirty-three percent (233%) and between three hundred percent (300%) to infinite translation depending on the ratio of the incorporated coplanar loop and the slopes of the variable ratio gear-sets incorporated.

According to one aspect of the invention there is provided an infinitely variable transmission comprising an input shaft and an output shaft. Power from the input shaft is split to produce at least two parallel streams: a fixed periodic output and a tunable periodic output. The fixed periodic output and the tunable periodic output are applied to a common control cage for outputting power to the output shaft. The control cage phase shifts the tunable periodic output relative to the fixed periodic output to vary the output ratio between zero and unity.

Description of the Drawings

The invention may best be understood with reference to the accompanying drawings wherein illustrative embodiments are shown:

Figure 1 shows a side cross sectional view of a variator cage/pinion type of a function generator incorporating a coplanar reverted gear-train-loop with a 2/1 ratio, together with the variable ratio gear-sets and the kinematic action of a twin, shadow cam lifter and associated passive coupling with a conical slipper-body interface;

Figure 2 shows a cross section and a perspective exploded view of a hydraulically controlled rotary actuator of the variator of Fig. 1, together with a plan view of the rotor/stator combination and a generated view of the land boundary of the pressure and exhaust domains on the outer cylindrical surface of the spool;

Figure 3 shows a perspective exploded view of the variator of Fig. 1;

Figure 4 are a series of graphs showing the ratios of relative rotation between driven/ driver variable ratio gear-sets with slopes of constant acceleration of 0.500, 0.375, 0.250 and 0.125 over a period of 220^0, followed by a sinusoidal return rotation over a period of 140^0, rotation, and a projected shape of respective pitch circles;

Figure 5 illustrates a method of generating teeth on the pitch circle profiles of variable ratio gear-sets as projected by the graphs of Fig.4, with teeth of constant-contact-ratio that result in teeth flanks of varying lengths as shown in Fig's. 5a and 5b;

Figure 6 are graphs showing the relative rotation of an annulus, a cage, a pinion and a passive coupling of a cage/pinion type function generator with oppositely orientated variable ratio gear-sets with slopes of acceleration, respectively of -0.500 and +0.500 over 220^0 of rotation followed by a cyclical sinusoidal return, as phase rotation of the driver gears is respectively shifted -45^0, -90^0 and -125^0;

Figure 7 are graphs similar to those of Fig.6, except that phase rotation of the driver gears is respectively shifted +45°, +90° and ÷125°.

Figure 8 is a cross sectional view of a variator with annulus/ pinion type function generators, a two gear-set rotary actuator to phase shift the common driver gears of the incorporated variable ratio gear-sets of different respective slopes, and the shadow cams of a twin, circular cam lifter with a passive coupling situated between an annulus of a coplanar-reverted-gear-train-loop of generator and a driven variable ratio gear of the variable ratio gear-set with a lessor slope of acceleration;

Figure 9 is a schematic of the variator of Fig.8 incorporating variable ratio gear-sets with slopes of acceleration of 0.500 and 0.250 respectively, the relative phase relationship of the incorporated variable ratio gear-sets and the twin, circular shadow cams lifter, together with a series of graphs depicting the relative rotation of a cage with constant input velocity, a pinion with attached common driven gear of a variable ratio gear-set with an acceleration slope of 0.500, a kinematically restrained annulus and passive coupling with attached driven gear of a variable ratio gear-set with an acceleration slope of 0.250 as phase rotation of the common driver gear is respectively shifted -45°, -90° and -125°;

Figure 10 is a schematic of the variator of Fig 11 with a coplanar reverted gear-train loop with a 3/2 ratio, the relative phase relationship of the incorporated variable ratio gear-sets and the twin, circular shadow cams lifter, together with a series of graphs depicting the relative rotation of a cage with constant input velocity, an annulus with attached common driven gear of a variable ratio gear-set with an acceleration slope of 0.375, a kinematically restrained pinion and passive coupling with attached driven gear of a variable ratio gear-set with an acceleration slope of 0.500 as phase rotation of the common driver gear is respectively shifted +45°, +90° and +125°;

Figure 11 shows a sectional view of a function generator of a annulus/pinion type variator, with variable ratio gear-sets in a same phase relationship and having different slopes of constant acceleration and a twin, shadow cam lifter similar to Fig.1, but, with a control actuator which in this case employs a flat, tapered radially coiled spring between the common driver gears of the variable ratio gear-sets to automatically control their phase relationship in proportion to the varying torque loads of translation through the variator;

Figure 12 is a schematic of the variator of Fig.11 with a coplanar reverted gear-train loop with a 2/1 ratio, the relative phase relationship of the incorporated variable ratio gear-sets and the twin, circular shadow cams lifter, together with a series of graphs depicting the relative rotation of a cage with constant input velocity, an annulus with attached common driven gear of a variable ratio gear-set with an acceleration slope of 0.250, a kinematically restrained pinion and passive coupling with attached driven gear of a variable ratio gear-set with an acceleration slope of 0.500 as phase rotation of the common driver gear is respectively shifted -45°, -90° and -125°;

Figure 13 is similar to Fig.12 except that phase rotation of the common driver gear is respectively shifted +45°, +90° and +125°;

Figure 14 is a schematic of the variator of Fig.11 with a coplanar reverted gear-train loop with a 3/2 ratio, the relative phase relationship of the incorporated variable ratio gear-sets and the twin, circular shadow cams lifter, together with a series of graphs depicting the relative rotation of a cage with constant input velocity, an annulus with attached common driven gear of a variable ratio gear-set with an acceleration slope of 0.375, a kinematically restrained pinion and passive coupling with attached driven gear of a variable ratio gear-set with an acceleration slope of 0.500 as phase rotation of the common driver gear is respectively shifted -45°, -90° and -125°;

Figure 15 is similar to Fig.14 except that phase rotation of the common driver gear is respectively shifted $+45^0$, $+90^0$ and $+125^0$;

Figure 16 is a schematic of the variator of Fig.11 with a coplanar reverted gear-train loop with a 3/2 ratio, the relative phase relationship of the incorporated variable ratio gear-sets and the twin, circular shadow cams lifter, together with a series of graphs depicting the relative rotation of a cage with constant input velocity, an annulus with attached common driven gear of a variable ratio gear-set with an acceleration slope of 0.250, a kinematically restrained pinion and passive coupling with attached driven gear of a variable ratio gear-set with an acceleration slope of 0.375 as phase rotation of the common driver gear is respectively shifted -45^0, -90^0 and -125^0;

Figure 17 is similar to Fig.16 except that phase rotation of the common driver gear is respectively shifted $+45^0$, $+90^0$ and $+125^0$;

Figure 18 is a sectional view of a variator with an annulus/pinion type function generator similar to Fig.11, with coplanar reverted gear-train-loop of ratio 4/3 and variable ratio gear-sets with slopes of 0.250 and 0.1875, that generates periods of constant veloity with amplitudes of +0.5 and -0.5 for an overall range of 3/1;

Figure 19 is a sectionalal view of a variator similar to Fig.8, but with a geared rotary actuator with an attached third reverted gear-sets that rotates a cam of the twin shadow cam lifter;

Figures 20 and 20a are sectional views of a variator cage/pinion type function generator, respectively incorporating a right angled sliding block arrangement and a rocker system for deactivating the passive couplings employed in the continuum;

Figure 21 is a sectional view of an infinitely variable all gear variator transmission showing the location of the main functional sub-assemblies;

Figures 22, 22a and 22b are sectional views of a regenerative coplanar reverted-gear-train-loop, a hi-lo coplanar reverted-gear-train-loop with an interactive spring/hydraulic selection mechanism with conical interfaced body and slipper passive couplings and a three dimensional exploded view of a variable displacement, lubricating and pressure disc pump;

Figure 23 is a side sectional view of an infinitely variable transmission the present invention;

Figure 24 is a schematic view of the power transmission of the embodiment of Figure 23,

Figure 25 is a graph illustrating in-phase rotation of the embodiment of Figure 23, using 220-220° identical 0.5 slope asymmetrical gearsets;

Figure 26 is a graph illustrating out-of-phase rotation of the embodiment of Figure 23;

Figure 27 is a graph illustrating in-phase rotation of the embodiment of Figure 23, using 180-180° identical symmetrical gearsets;

Figure 28 is a graph illustrating out-of-phase rotation of the embodiment of Figure 23;

Figure 29 is a schematic view of the power transmission of a further embodiment of the infinitely variable transmission of the present invention;

Figure 30 is a graph illustrating in-phase rotation of the embodiment of Figure 29, using 220-220° identical 0.5 slope asymmetrical gearsets;

Figure 31 is a graph illustrating out-of-phase rotation of the embodiment of Figure 29;

Figure 32 is a graph illustrating out-of-phase rotation of a third embodiment of the present invention using a 2/1 differential and passive couplings;

Figure 33 is a graph illustrating in-phase rotation the embodiment of Figure 29;

Figure 34 is a side sectional view of a bevel to bevel embodiment of the present invention;

Figure 35 is a side sectional view of a bevel to carrier function generator embodiment of the present invention; and

Figure 36 is a side sectional view of a bevel to carrier function generator embodiment of the present invention.

Description of the Invention

Referring to Figure 23, an infinitely variable transmission 10 of the present invention is illustrated. The transmission 10 generally comprises a casing 12, having an oil pan 14, an input shaft 16 and an output shaft 18. A flywheel assembly is mounted onto input shaft 16 and receives drive from an engine, not illustrated. Also mounted on the input shaft 16 is an oil pump 22 which delivers oil to the various components of the transmission 10. For simplicity purposes, the oil delivery paths have not been illustrated. However, it will apparent to those skilled in the art how to deliver oil to the requisite components.

Input shaft 16 is journal mounted in the casing 12 at bearings 24, 26 and extends generally axially of the casing 12. Reduction gearset 28 splits the power from the input shaft 16 to two diametrically opposed parallel power transfer shafts 30, 32. The two shafts 30, 32 are counter rotating to offset vibrations. The components mounted on each shaft 30, 32 are identical and therefore only one of the shafts will be described.

The gearset 28 preferably reduces the speed of rotation of the shaft 30 to about half the speed of rotation of the input shaft 16. Shaft 30 is journal mounted at bearings 34, 36. Mounted along shaft 30 is a bevel gearset 38, a stationary driven gearset 40 and a tunable gearset 42. Bevel gearset 38 splits the power into two parallel power streams. In the illustrated embodiment, bevel gearset 38 is differential type of gear-set which drivingly

rotates gear-sets 40 and 42. Gearsets 40 and 42 are mounted on separate collars so that each gearset 40, 42 can rotate independently of each other The mating half of the gearsets 40, 42 are also mounted on collars 44, 46 which are also able to rotates independently of each other. In this embodiment, the collars 44, 46 are coaxially mounted about input shaft 16.

Gear-sets 40, 42 are preferably non-circular gears having constant contact. The asymmetrical design of the gear-sets 40, 42 will generate a pulsating output rotational speed from a constant input rotational speed. Alternatively, identical symmetrical gear-sets may also be used.

Collars 44, 46 are each geared to the control cage 48 having a geared shaft 50 which responsively rotates from input from collars 44, 46. Rotation of shaft 50 will responsively cause driving rotation of the control cage 48. The outer housing 52 of control cage 48 is coupled to a regenerative co-planar loop gearset 54 via coupling 56. The output of gearset 54 is coupled to a clutch assembly 58. The output of clutch assembly 58 is coupled to a hi-lo coplanar loop gearset 60 which in turn drivingly rotates output shaft 18 at a desired output speed.

Gearset 42 is mounted on a tunable mount assembly 62 Responsive to external input such as a controller or a manual mechanism, gearset 42 can be caused to rotate in phase with gearset 40 or out of phase therewith. Preferably, assembly 62 rotates gearset 42 up to about 110 of phaseshift.

Referring Figure 24, the power transfer is schematically illustrated. The input is first reduced in speed. The power is then split into parallel paths between gears *g* and *r*, which correspond to gearsets 40, 42, respectively. The parallel streams are then recombined to an output.

In response to the tunable mount assembly 62, the relative phase of gearset g can be shifted relative to gearset r. The kinematic constraint on the cage 48 is:

$$\dot{\omega}_c = (\dot{\omega}_r + \dot{\omega}_g)/2$$

When translating through the bevel gear-sets to the fixed ratio gearset, the translation ratio varies from 1/1 to 1.25/1 or 25%, and in reverse translation in the opposite sense, the ratio varies from 1/1 to 1/(1-(1.25-1)) or 33%. When the casing spool is rotated as the input member and the fixed gearset is a stationary reactive component, output varies from 0 to (1.25-1) and 1/((1-(1.25-1)-1) so that the transmission has an infinite ratio. Centrifugal loading on the rotating function generators is a limitation of the present configuration. Spiroidal one-way clutches can also be located between the bevel gearset 38 and a driven gearset 40, 42 without changing the kinematics of the present invention.

Other mechanisms for dividing the input power torque to a parallel path include a bevel to carrier function generator. The schematic of the power transfer is illustrated in Figure 29. Bevel to carrier function generators use gearsets 40,42 which have different slopes, where slope is defined as one-half the amplitude of linear acceleration of the driven gear. The kinematic relationship of the components is a function of the ratio of the differential. Although applicable to bevel to bevel type transmissions, as previously described, an increase in differential ratio decreases the amplitude of the periods. Bevel to carrier function generators with a 1/1 ratio differential and gearsets with slopes of 0.5 and 0.25 are capable of generating period amplitudes of from 1/1 to 1.5/1 or 50% to 1+(1/1.5) or 67%. With a differential ratio of 2 and a gearset slope of 0.5 and 0.33, by phase shifting in each direction, the amplitude of the period is +1 or -1. Consequently, with passive couplings, translated output will vary from 0 to 2. Again a casing spool input configuration is feasible with a bevel to carrier function generator. However, a fundamental consideration

of any variator is the tooth loading which favors a bevel to bevel configuration as previously described with the attendant sacrifice of amplification.

In the embodiments illustrated, two diametrically opposed parallel power transfer shafts 30, 32 have been illustrated. It is now apparent to those skilled in the art that any number of pairs of shafts may be used. Preferably, two pairs of diametrically opposed counter-rotating shafts are used.

It thus will be seen that the objects of this invention have been fully and effectively accomplished. It will be realized, however, that the foregoing preferred embodiment of the present invention has been shown and described for the purposes of illustrating the structural and functional principles of the present invention and is subject to change without departure from such principles.

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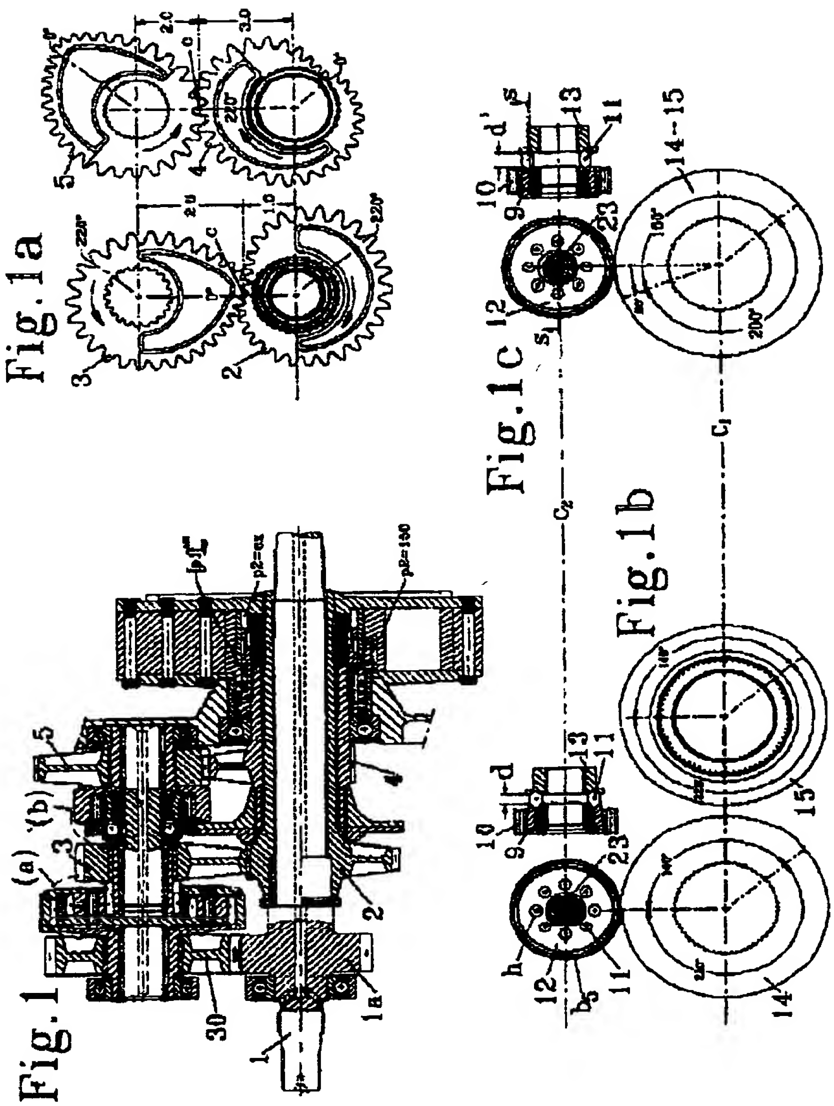

Fig.1 Fig.1a Fig.1b Fig.1c

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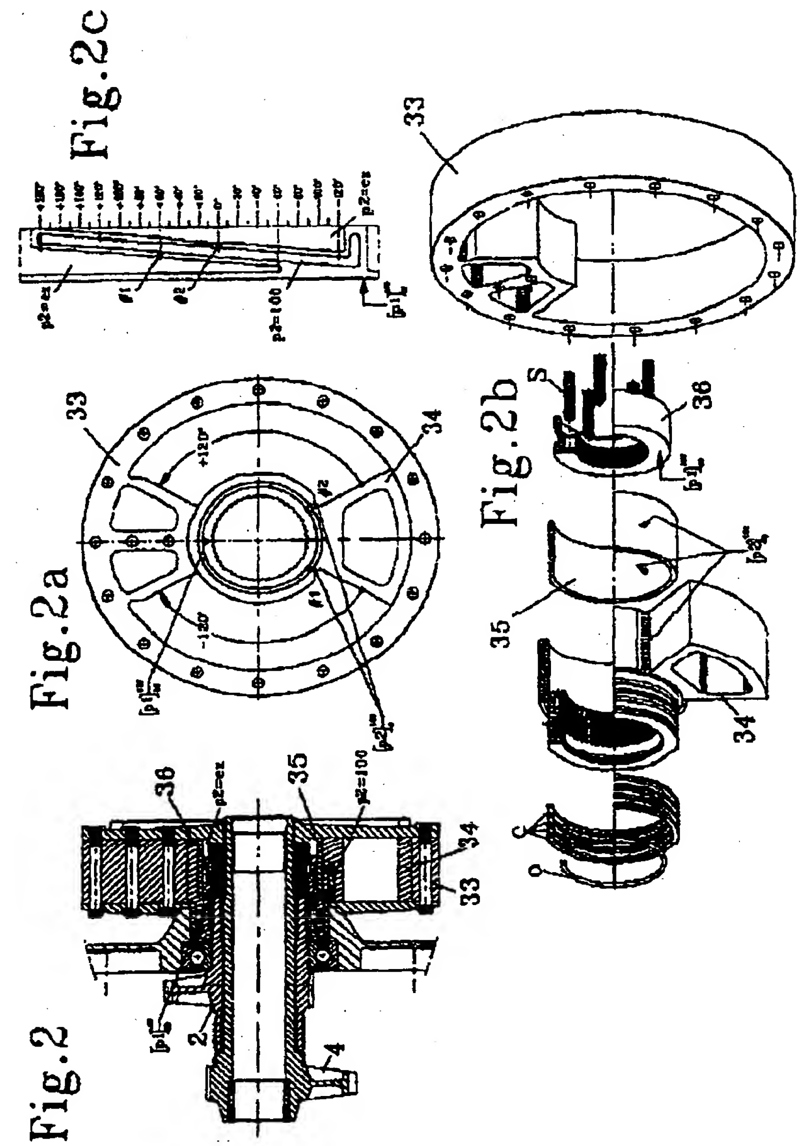

Fig.2 Fig.2a Fig.2b Fig.2c

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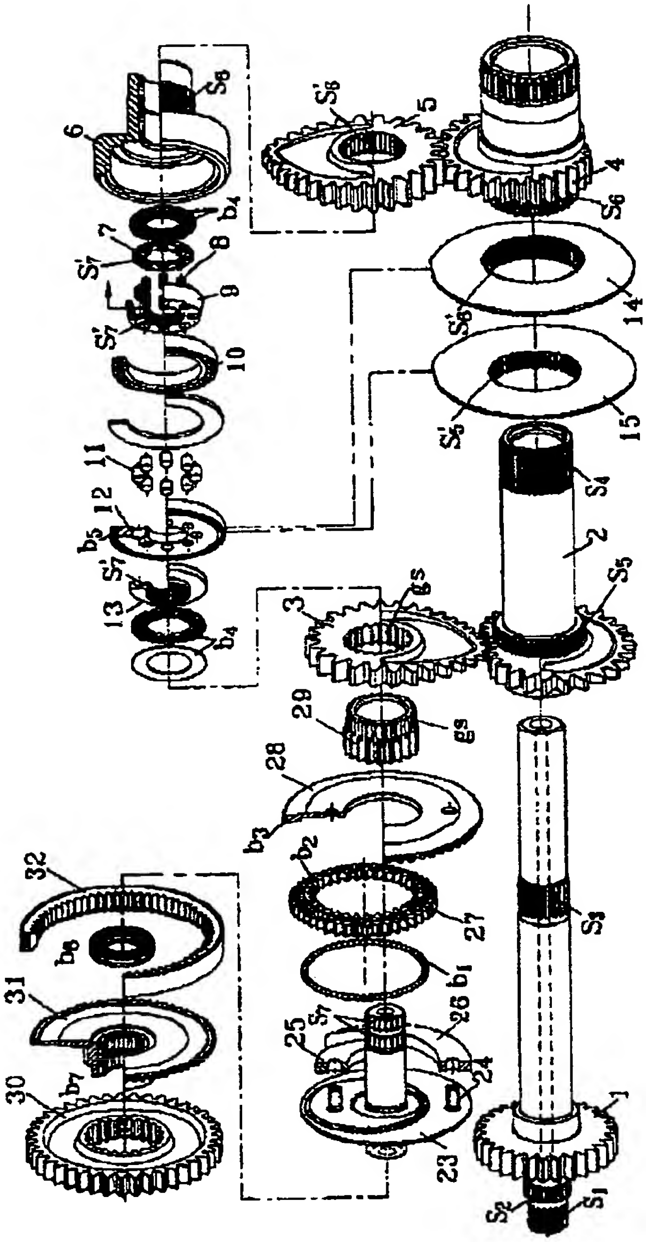

Fig.3

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Fig.4

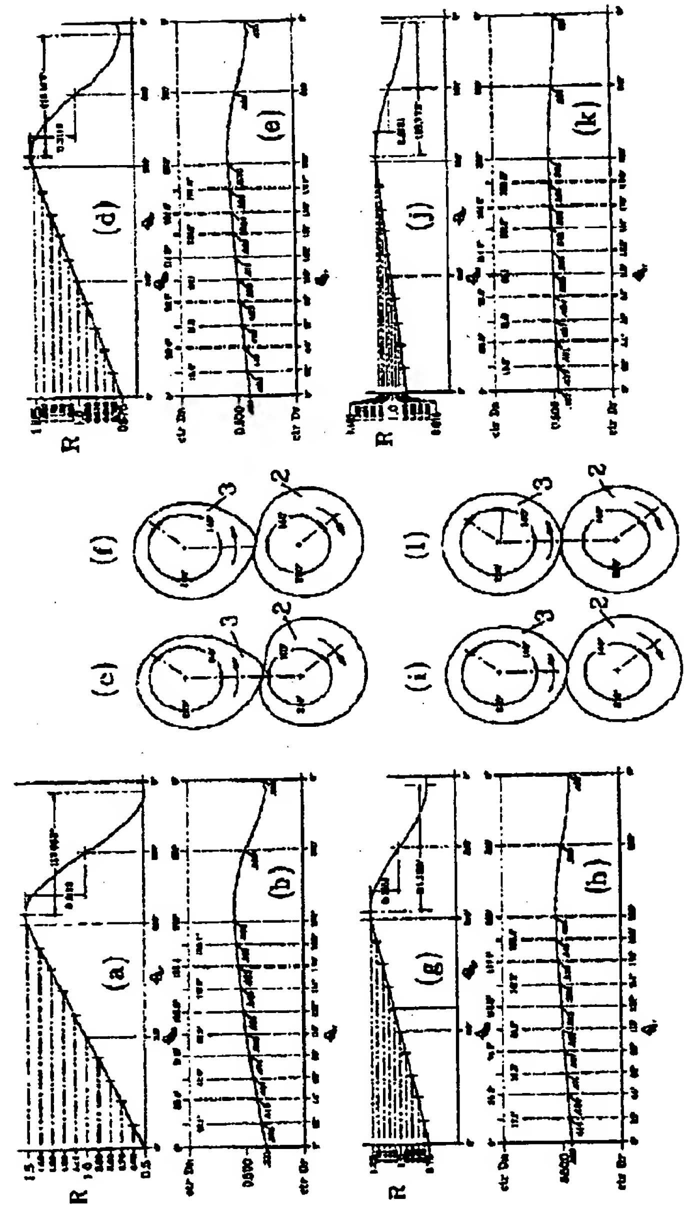

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Fig.5

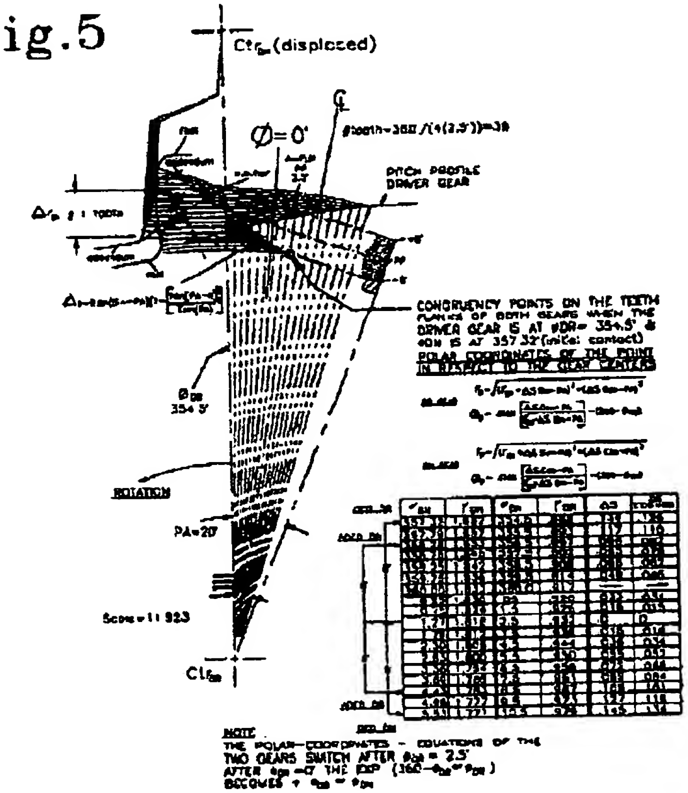

Fig.5a

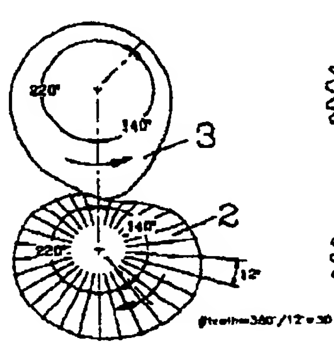

Fig.5b

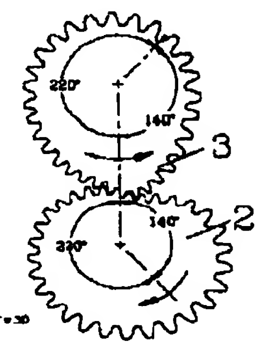

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Fig.6

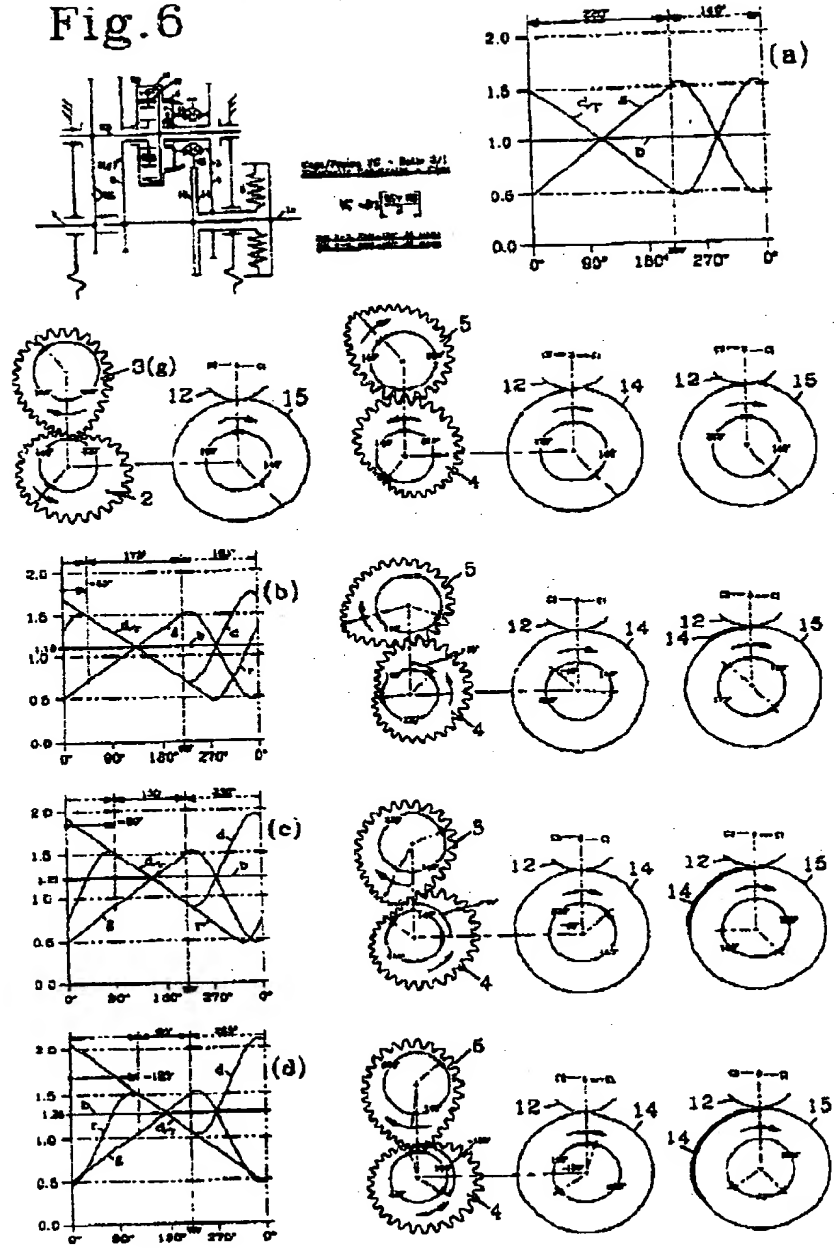

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Fig.7

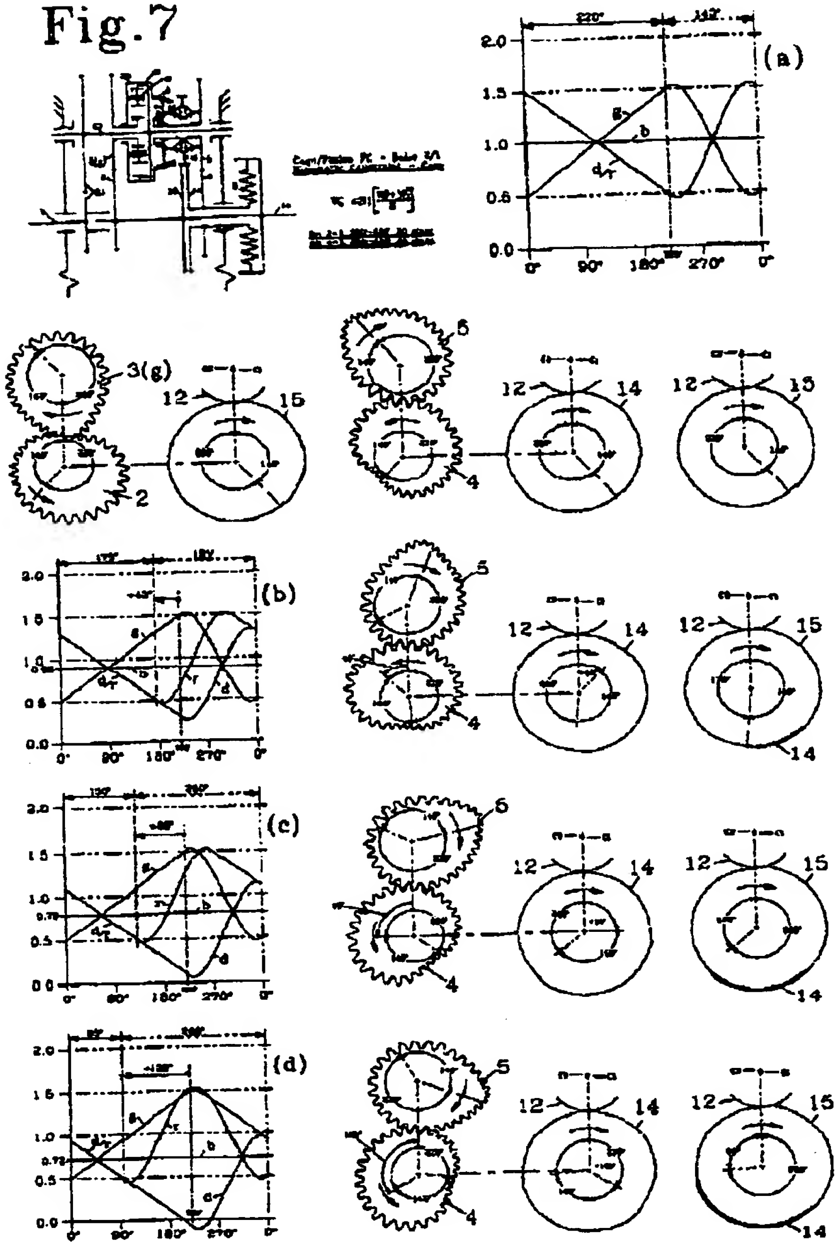

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Fig.8

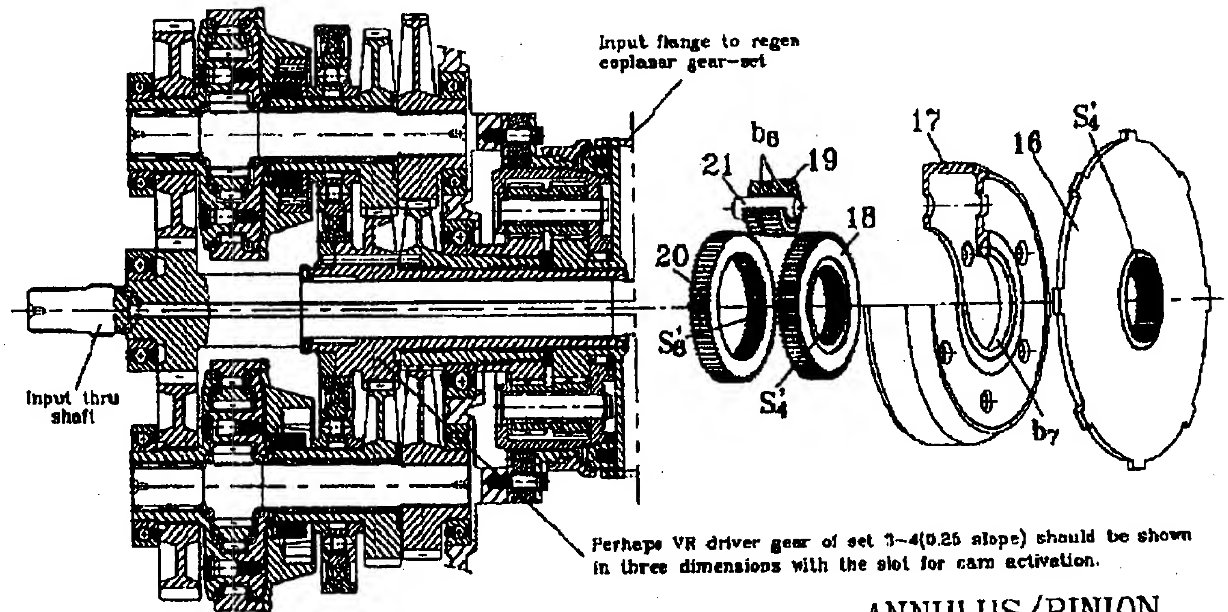

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Fig.9

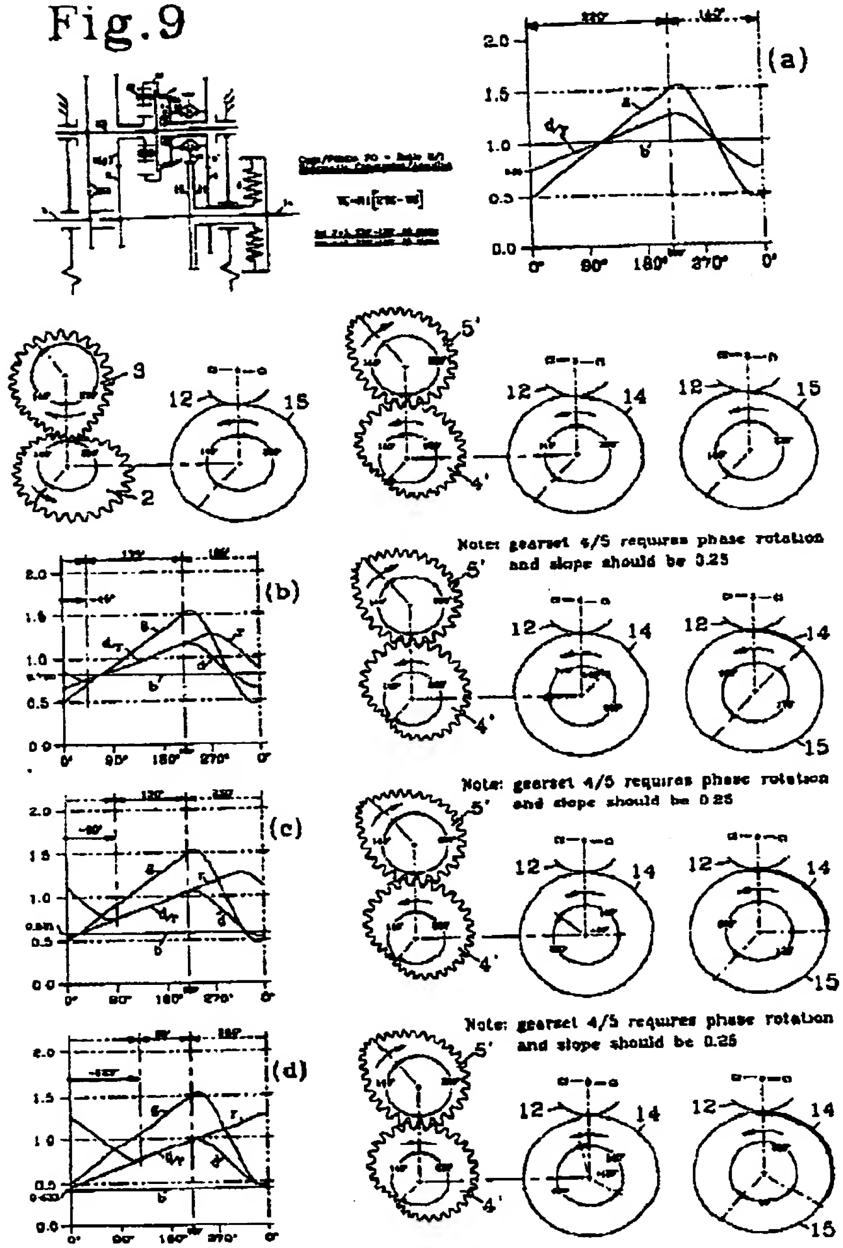

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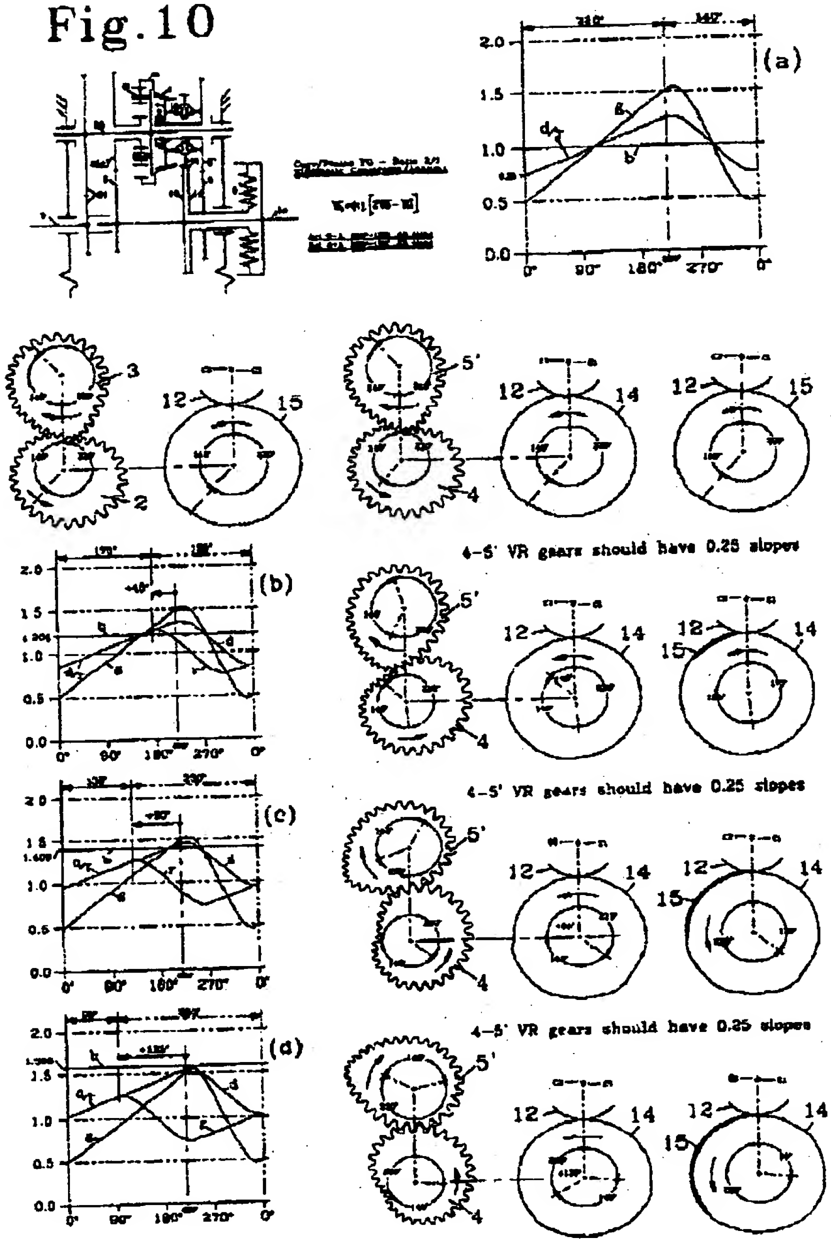
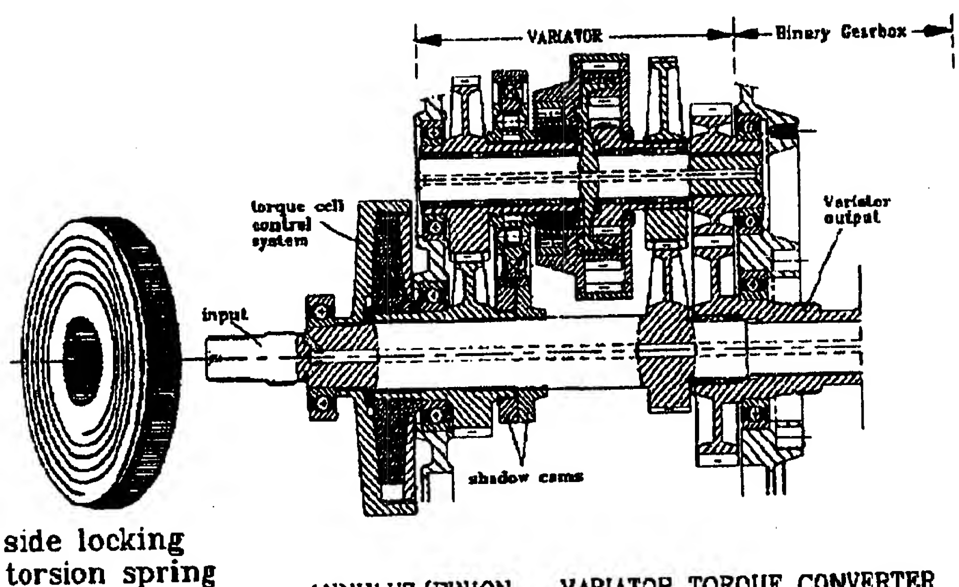
Fig.10



EXPRESS MAIL #EM494001309US

Fig.11

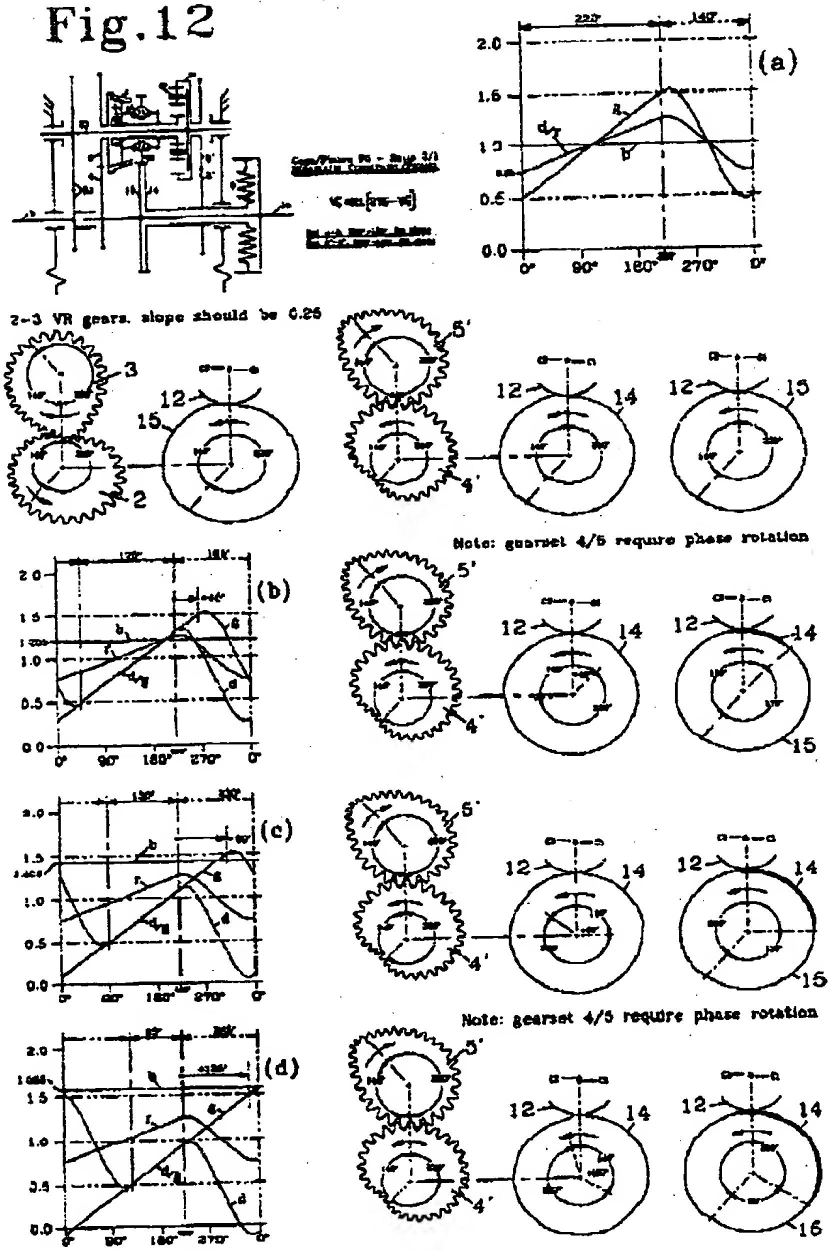


ANNULUS/PINION – VARIATOR TORQUE CONVERTER

Coplanar Loop ratio k =.167, VR-Gears with slopes of .5 and .375 output 1.00 ±1.00, for an infinite ratio change(zero output)

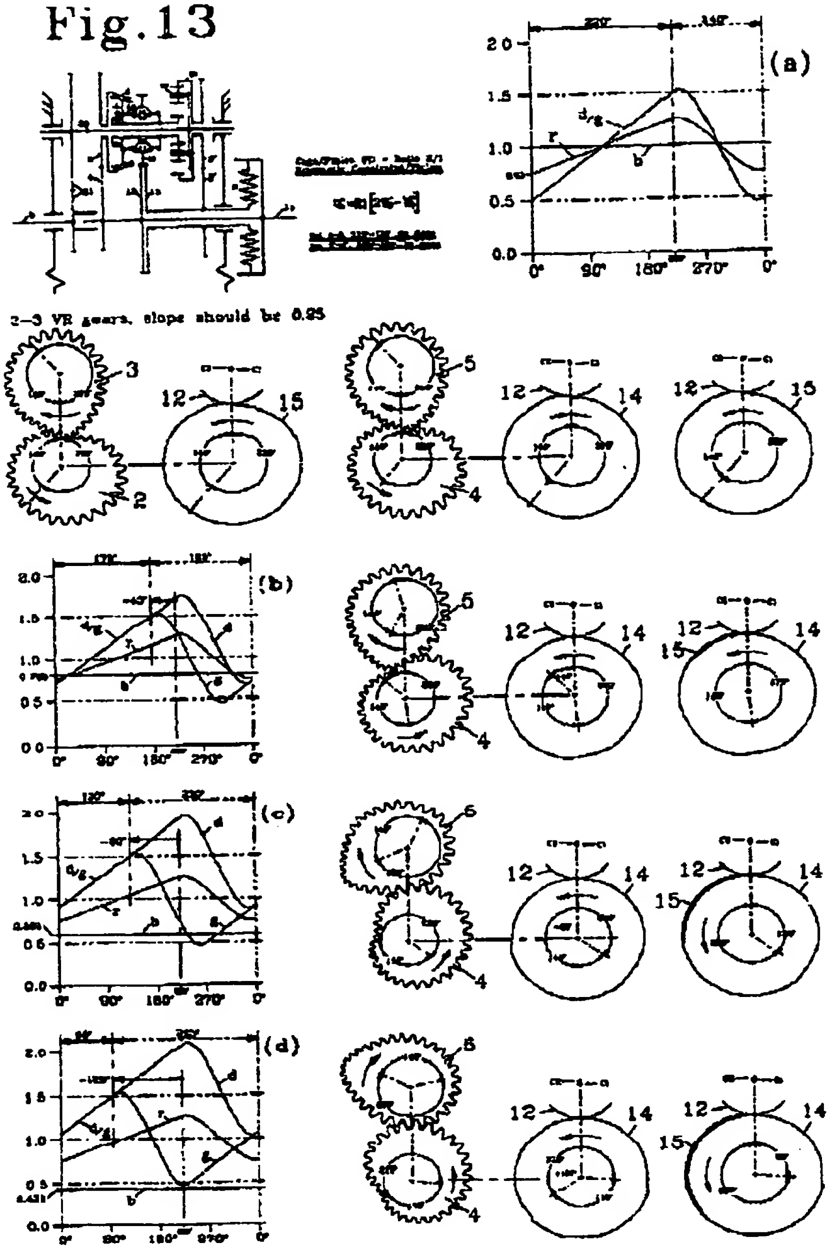
EXPRESS MAIL 494001309US

Fig.12



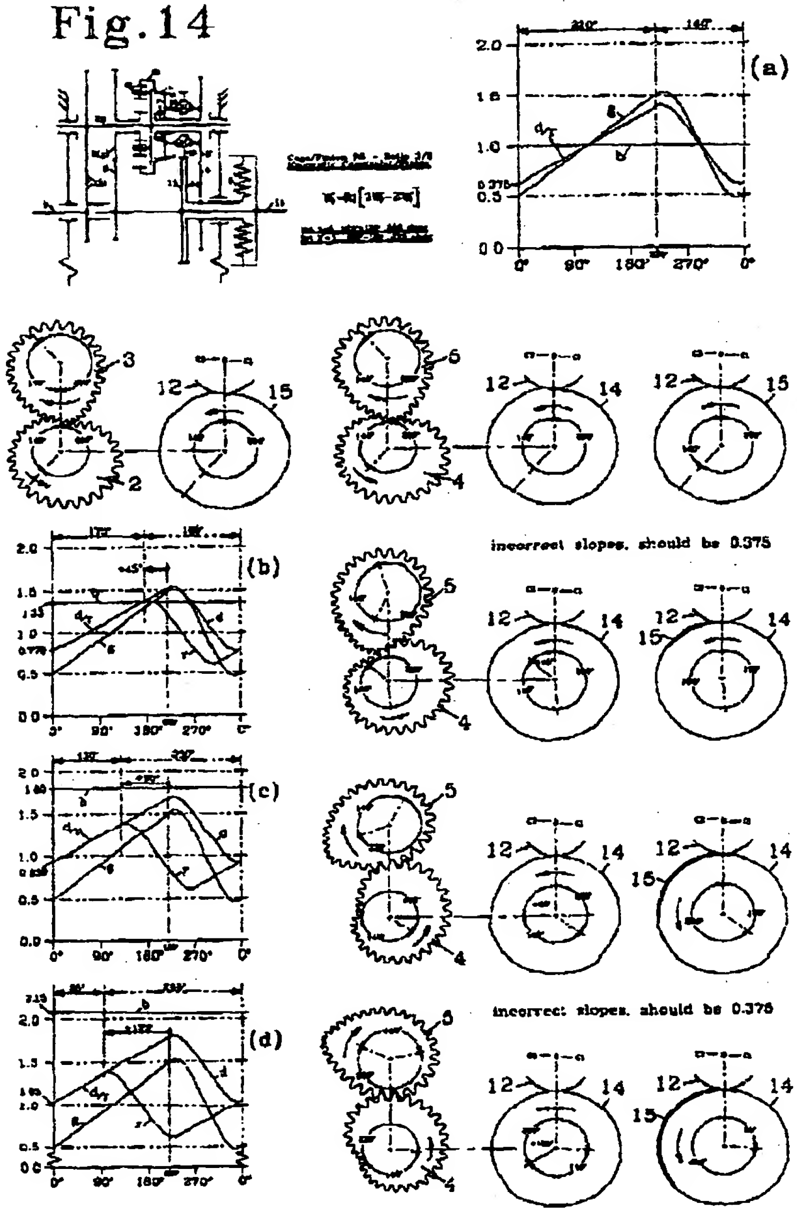
EXPRESS #EM494001309US

Fig.13



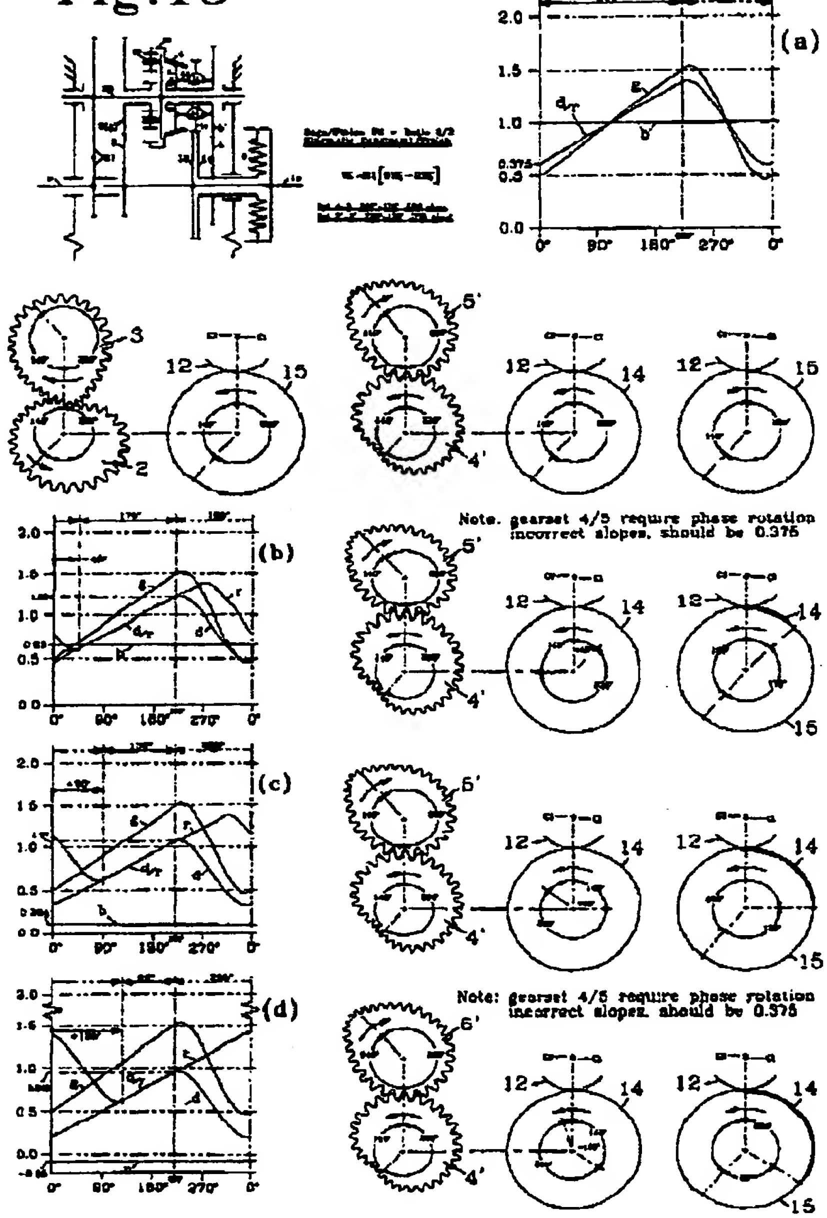
EXPRESS MAIL [illegible] 494001309US

Fig.14



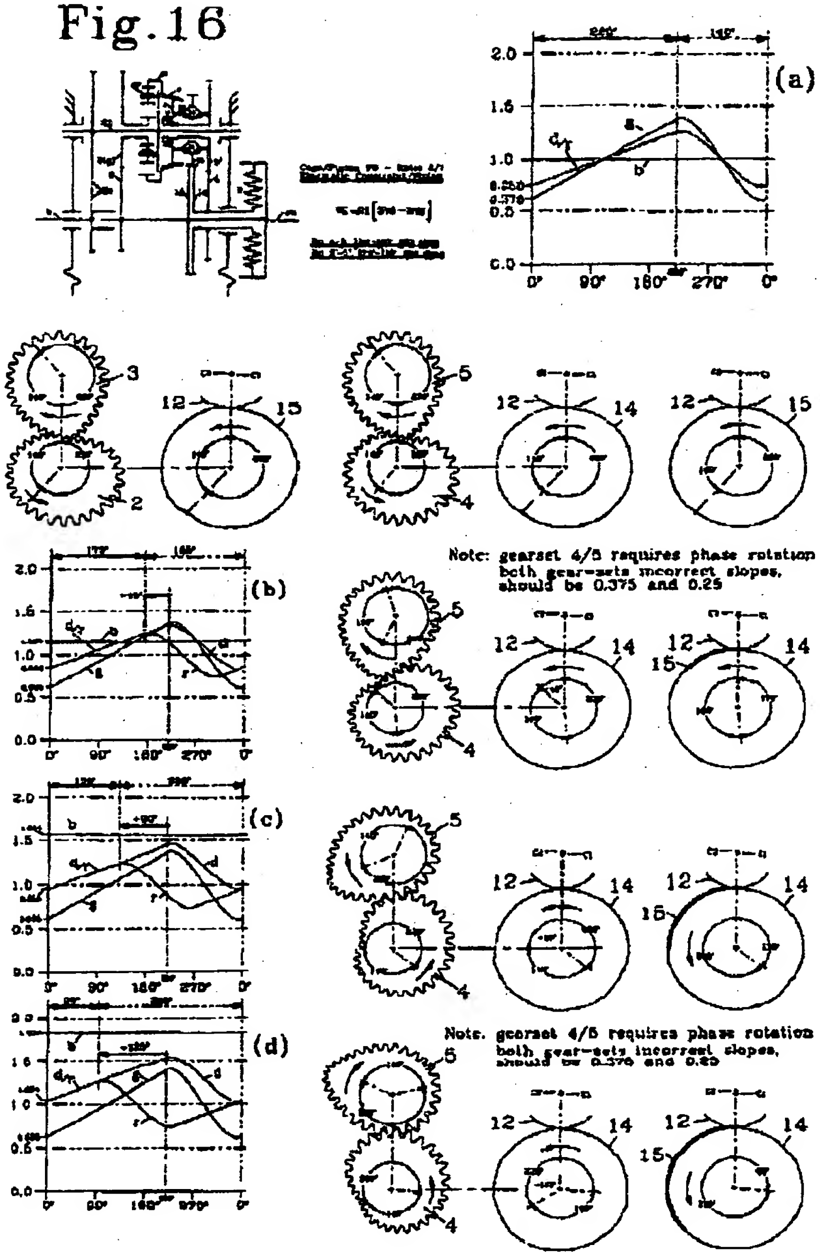
EXPRESS MAIL #EM494001309US

Fig.15



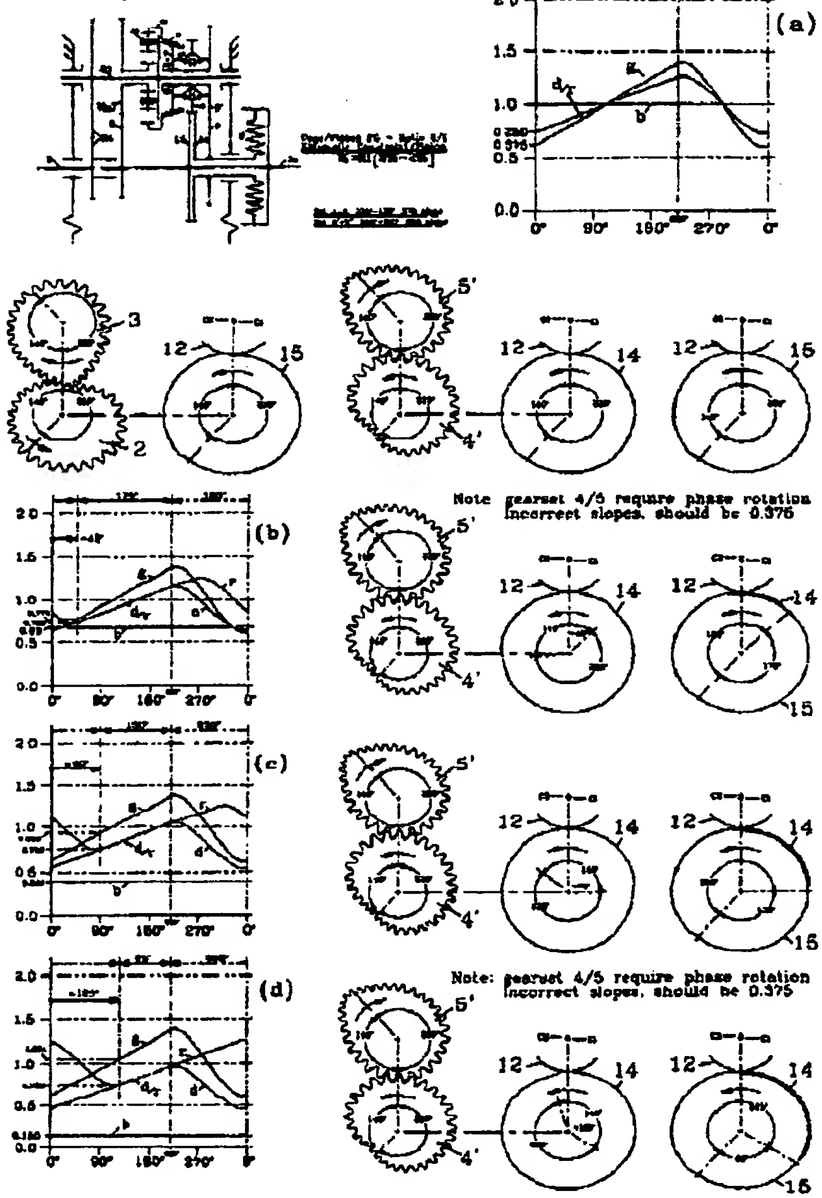
EXPRESS MAIL 94001309US

Fig.16



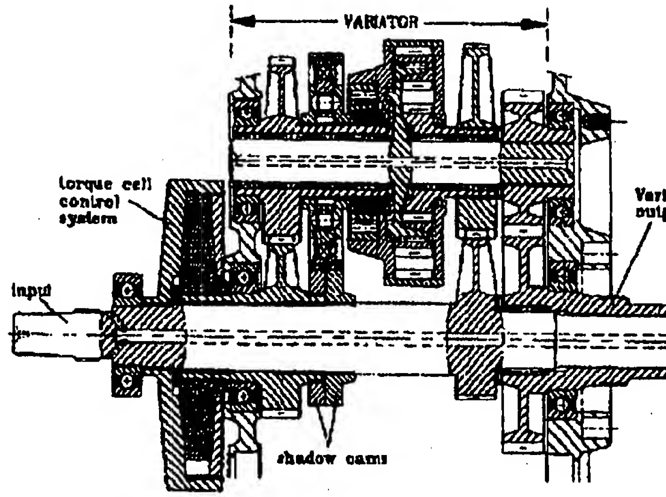
EXPRESS MAIL#EM494001309US

Fig.17



EXPRESS MAIL #EM494001309US

Fig.18



Coplanar Loop ratio 3/2 with VR-Gears with slopes of .25 and .1875 that result in amplitude change of +-.50

Pitch circle profiles @ .25 and .1875 slopes

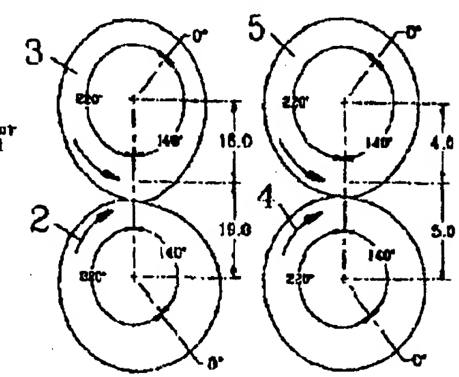
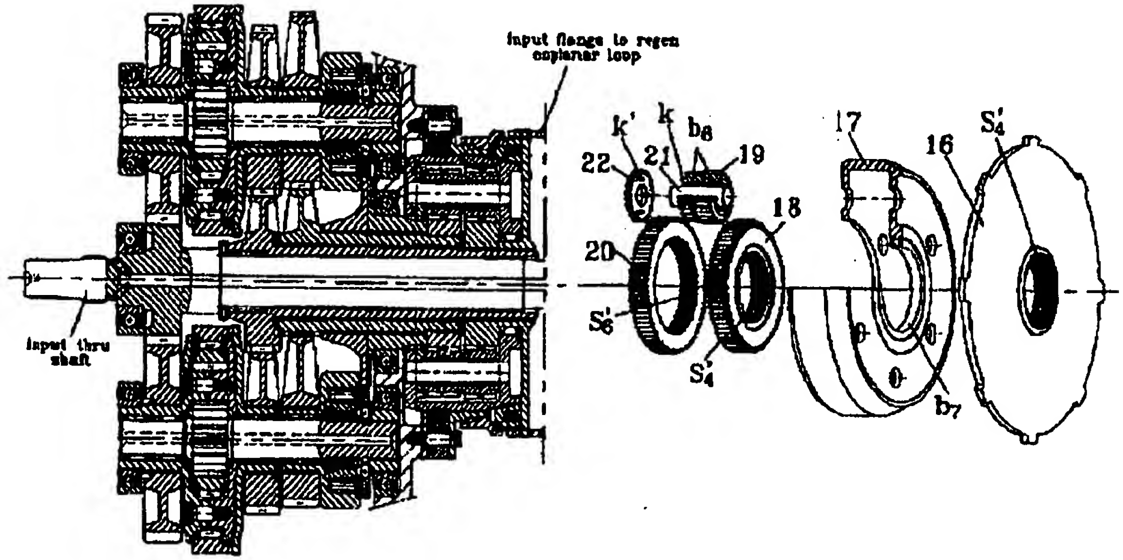


Fig.18a

ANNULUS/PINION
(BEVEL/CARRIER)

EXPRESS M[illegible] #EM494001309US

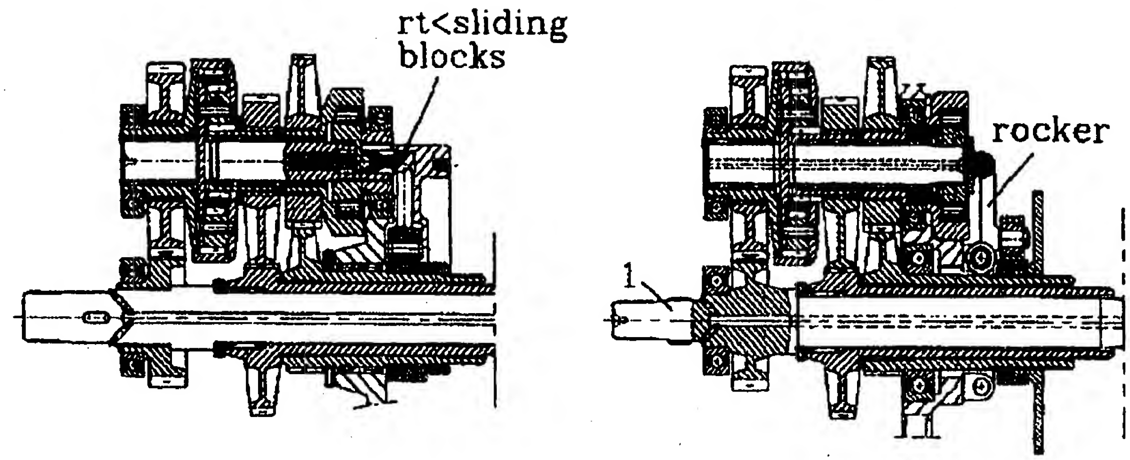
Fig.19



ANNULUS/PINION
(BEVEL/CARRIER)

EXPRESS MAIL 494001309US

Fig.20



CAGE/PINION
(BEVEL/BEVEL)

EXPRESS MAIL #EM494001309US

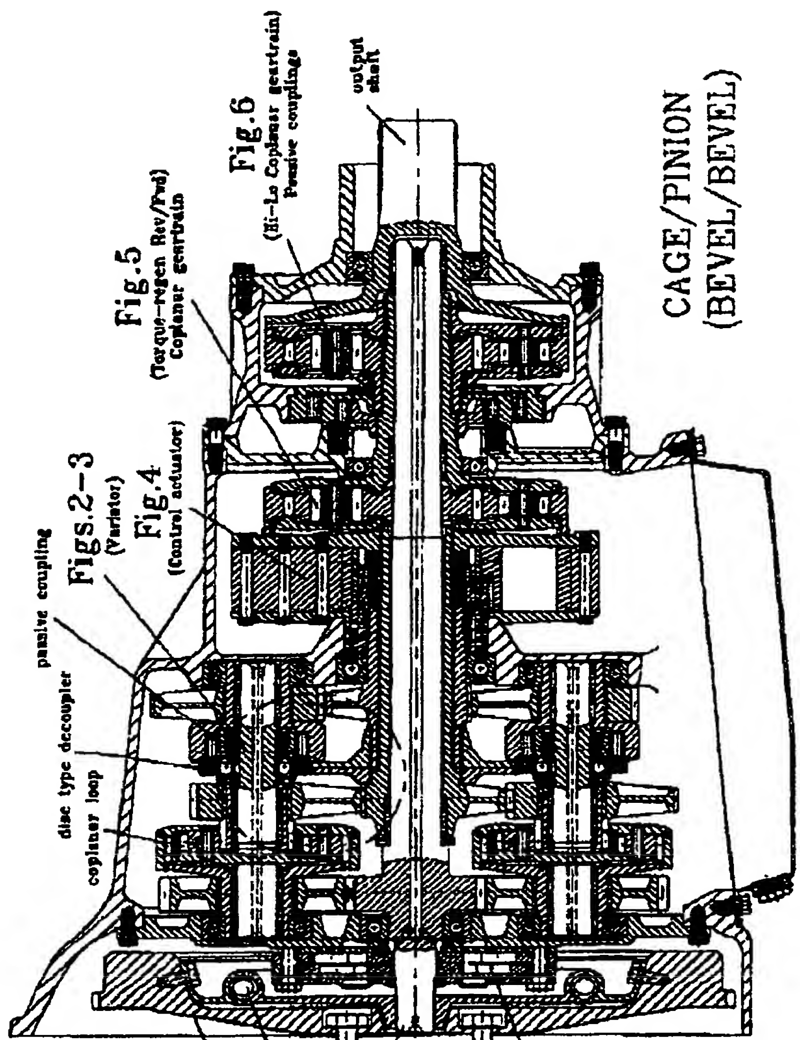


Fig.21

EXPRESS MAIL #EM494001309US

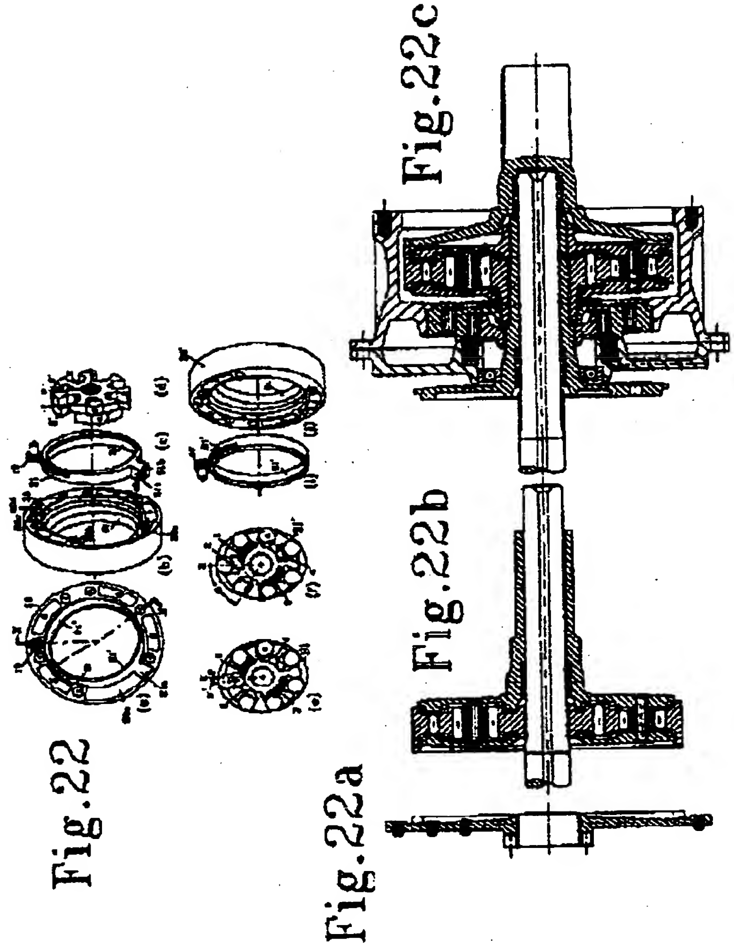


Fig.22

Fig.22a

Fig.22b

Fig.22c

EXPRESS MAIL #EM494001309US

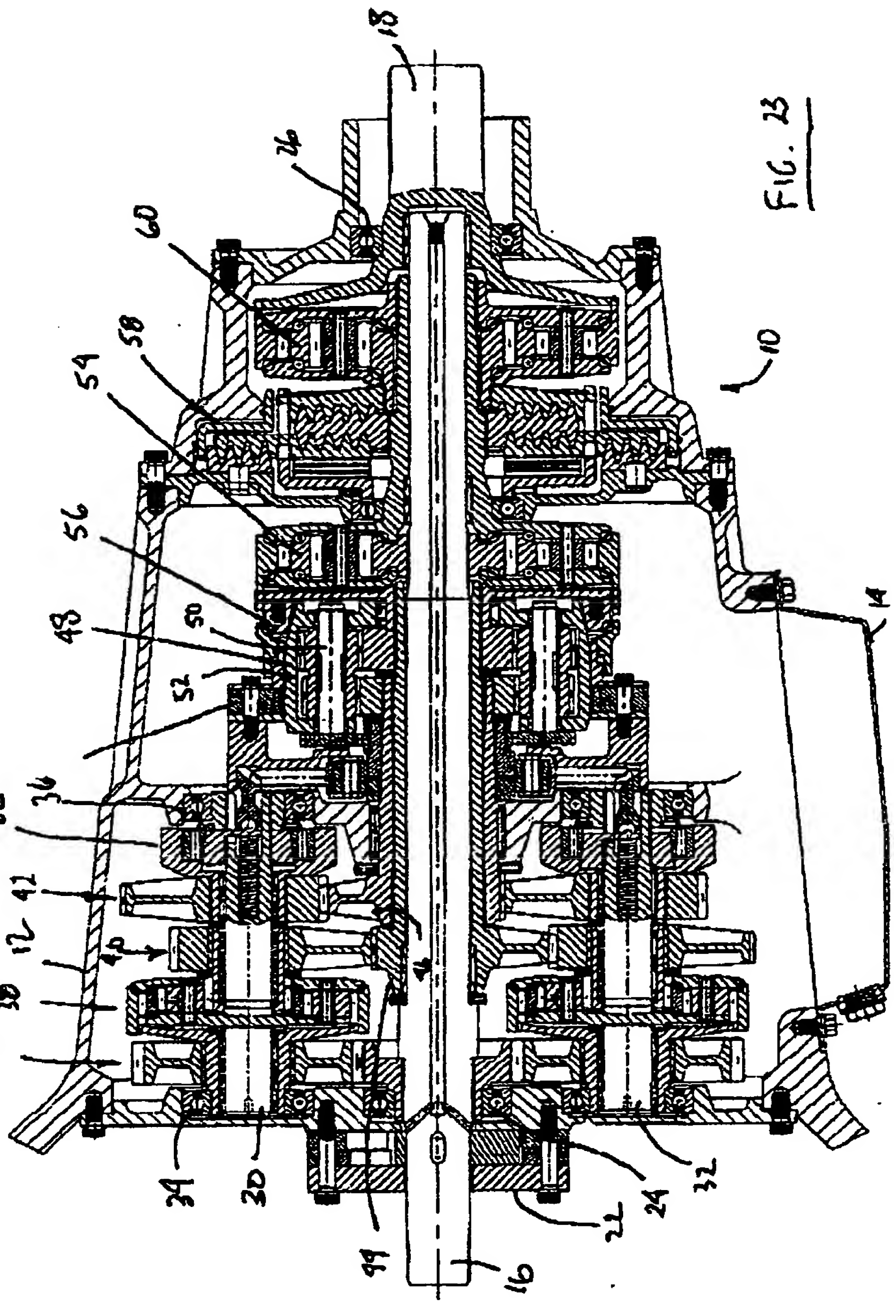


FIG. 23

EXPRESS MAIL #EM494001309US

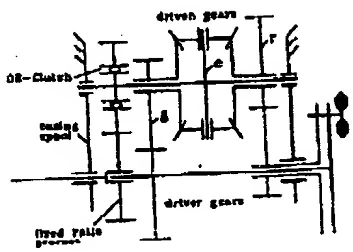


FIG. 24

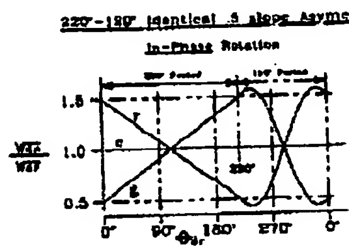


FIG. 25

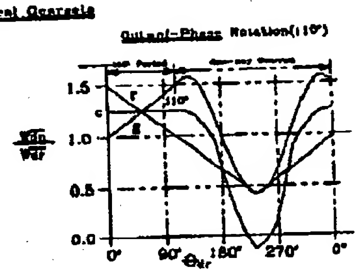


FIG. 26

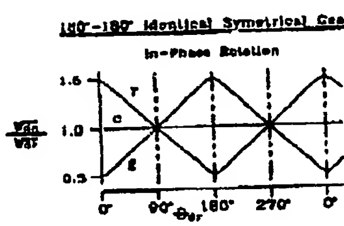


FIG. 27

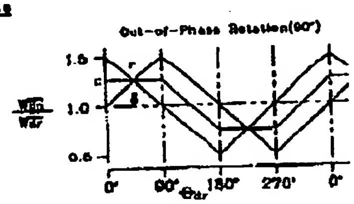


FIG 28

EXPRESS MAIL #EM494001309US

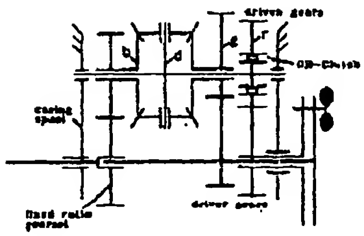


FIG 29

220°-120°, .5 & .25 slope Asymmetrical Gearsets

In-Phase Rotation 1/1 differential

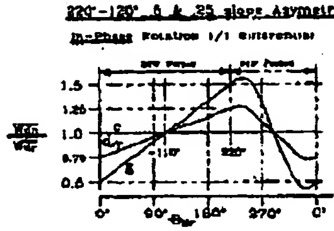


FIG 30

Out-of-Phase Rotation(110°)

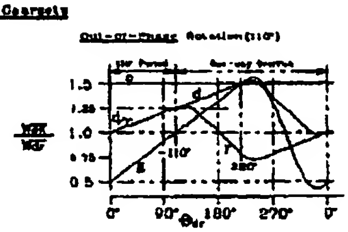


FIG. 31

Out-of-Phase Rotation(±110°) & 2/1 differential and passive couplings

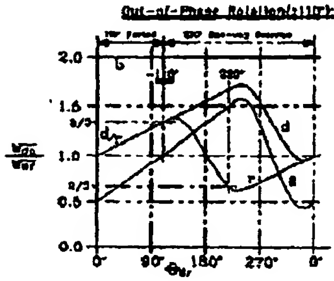


FIG. 32

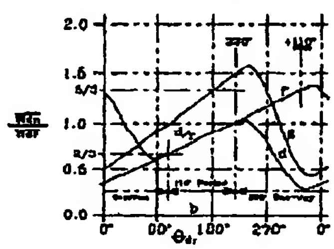


FIG. 33

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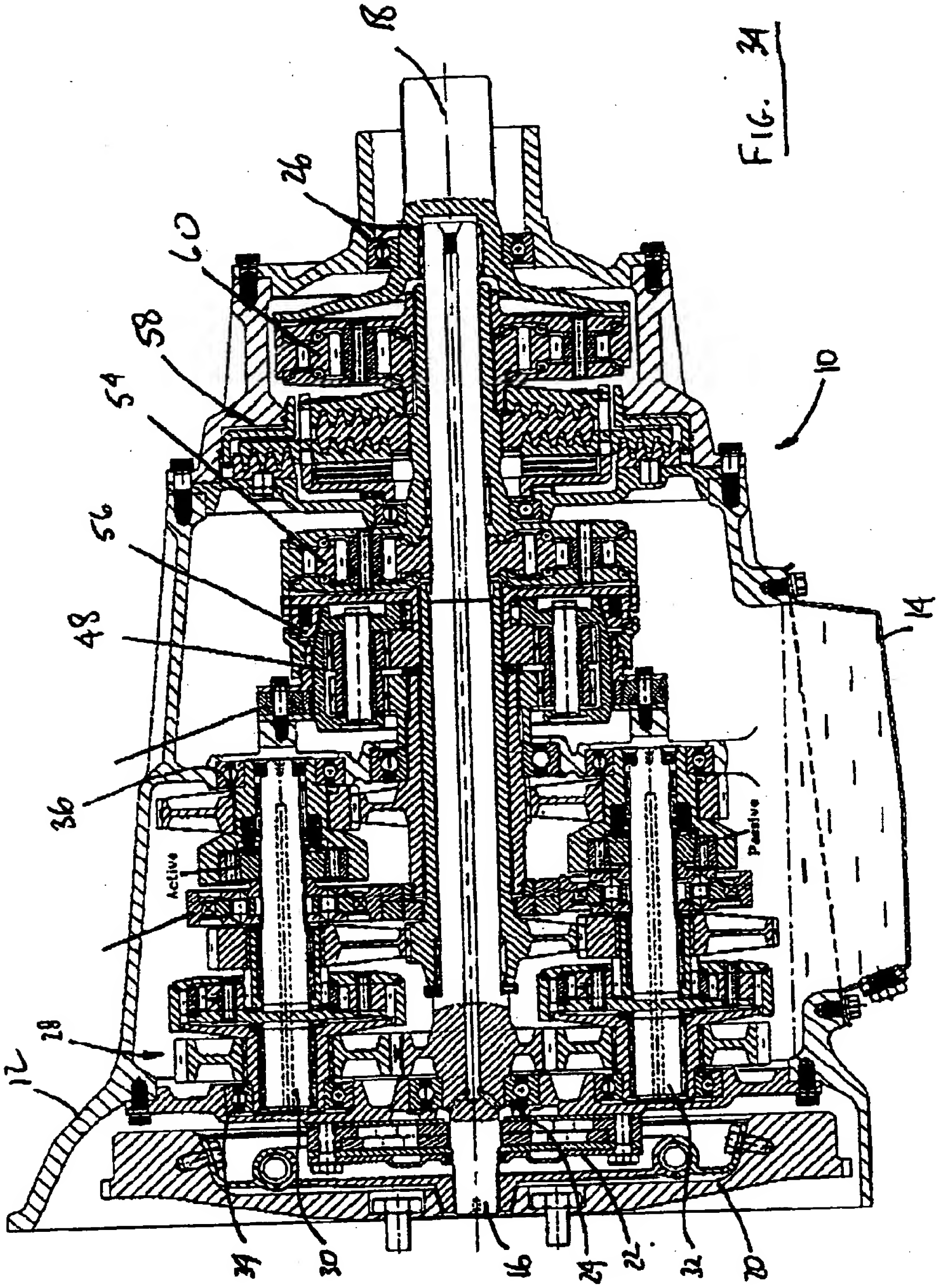

FIG. 3A

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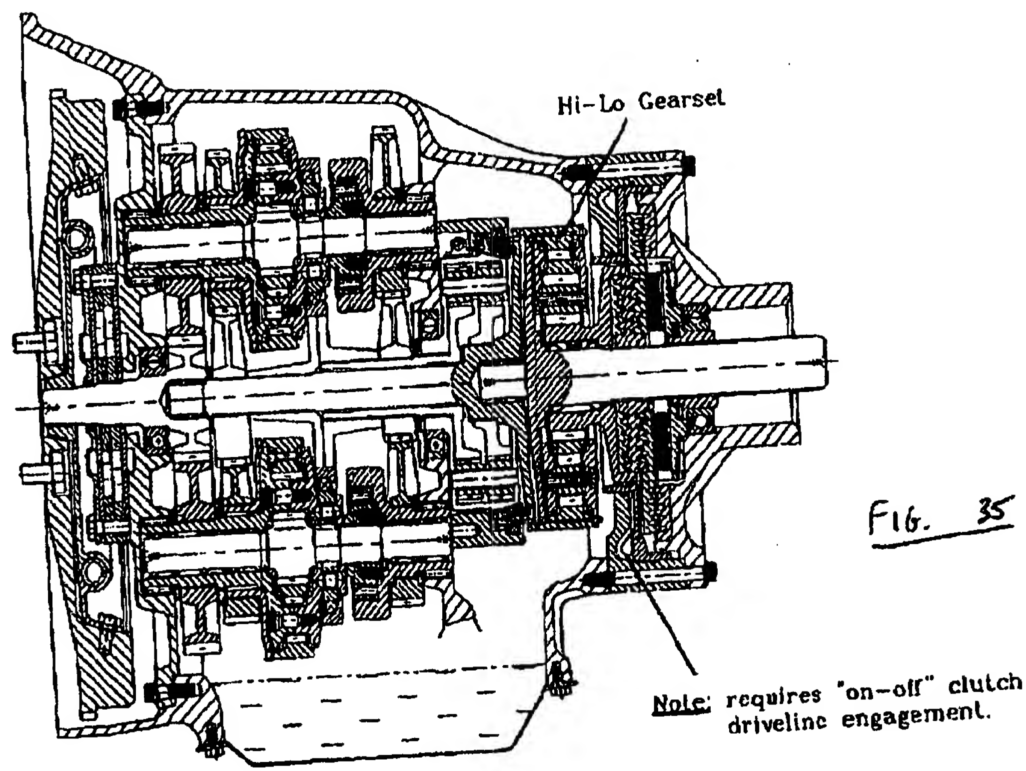
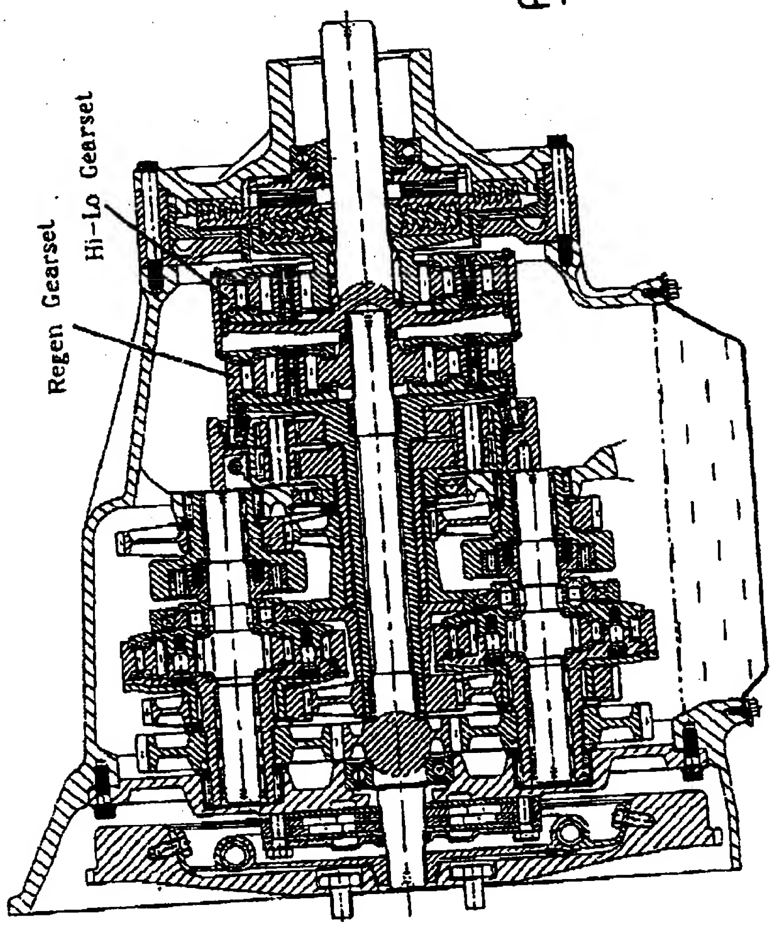


FIG. 35

EXPRESS MAIL #E[illegible]4001309US

FIG. 36



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